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ANALYTICAL INVESTIGATION OF RECIRCULATION-PUMP AND RADIATOR-AREA REQUIREMENTS FOR FLASH VAPORIZATION IN A TURBOELECTRIC SPACE POWER SYSTEM

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ANALYTICAL INVESTIGATION OF RECIRCULATION-PUMP AND RADIATOR-AREA REQUIREMENTS FOR FLASH VAPORIZATION IN A TURBOELECTRIC SPACE POWER SYSTEM

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SUMMARY

An investigation was made to determine the effect of heater-exit temperature, turbine-inlet temperature, turbine-exit to -inlet temperature ratio, fluid, and system-pressure loss on the recirculation-pump-power and radiator-area requirements for a modified Rankine power cycle employing flash vaporization.

For each fluid, turbine-inlet temperature, and system-pressure loss, there is an optimum value of heater-exit temperature that yields a minimum recirculation-pump power. Minimum recirculation-pump power increases with an increase in turbine-inlet temperature, turbine-exit to -inlet temperature ratio, or system-pressure loss. A sodium system requires the lowest recirculation-pump power, potassium higher power, and rubidium the highest.

For each fluid and turbine-inlet temperature there is an optimum value of turbine-exit to -inlet temperature ratio that yields a minimum specific radiator area. There is also an optimum flash-temperature drop that yields a minimum specific radiator area. Specific radiator area decreases with an increase in turbine-inlet temperature.

For certain operating conditions and fluids presently being considered for turboelectric space power systems, the required power output and radiator area for a modified Rankine cycle employing flash vaporization are not greatly in excess of those required for a cycle employing a boiling heat exchanger. However, the feasibility of applying the flash-vaporization technique to turboelectric space power systems cannot be determined from this study alone. A complete system optimization study must be made and the results compared with similar studies made for systems employing forced-convection boiling and pool boiling.

INTRODUCTION

Turboelectric power systems are currently being studied for application in space vehicles. A number of analyses (e.g., refs. 1 and 2) have shown the advantages of a Rankine (vapor-liquid) cycle over a Brayton (gas) cycle. The alkali metals appear to be the most attractive working fluids for a Rankine cycle operating in outer space.

Vaporization of the working fluid is a necessary part of a Rankine power system. The three major vaporization techniques are pool boiling, forced-convection boiling, and flash boiling. The negligible gravity forces in outer space necessitate the presence of an artificially induced force to ensure contact of the liquid phase with the heating surface. Consequently, the use of pool boiling would probably require some novel arrangement in order to satisfy this condition. Two-phase flow with vaporization, as encountered in forced-convection boiling, is a complex process, and a completely satisfactory model for the heat transfer and the fluid dynamics of such a system has yet to be evolved. Available information indicates, however, that a significant pressure drop may be required, especially in the case of complete vaporization, to overcome friction (ref. 3), supply the necessary increase in momentum (ref. 3), and ensure stable flow (ref. 4). The lack of basic design data for forced-convection boiling systems has led some investigators (e.g., ref. 2) to consider the flash-vaporization technique.

The use of flash vaporization permits the working fluid to remain in a liquid state within the heater and, therefore, assures continuous contact of liquid with the heat-transfer surface. A major disadvantage of the flash-vaporization technique is the vaporization of only a small amount of the liquid circulating in the flash loop, which may cause the circulation rate in the flash loop and the power requirement for the recirculation pump to become large. The problem of separating liquid from vapor is recognized but neglected in this study.

This analytical investigation was undertaken to determine the effect of fluid, turbine-inlet temperature, turbine-exit to -inlet temperature ratio, heater-exit temperature, and system-pressure loss (frictional loss plus boiling-suppression head) on the recirculation-pump power and radiator area for a modified Rankine power cycle employing flash vaporization. Sodium, potassium, and rubidium were investigated as working fluids, with turbine-inlet temperature ranging from 1760° to 2960° R and heater-exit temperature ranging from 0° to 400° R above turbine-inlet temperature. Turbine-exit to -inlet temperature ratio was varied from 0.6 to 0.9 and system-pressure loss was varied from 1 to 20 pounds per square inch.

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SYMBOLS

A_R	radiator area, sq ft
C_p	heat capacity, Btu/(lb)($^{\circ}$R)
ΔH_{vap}	heat of vaporization, Btu/lb
h	specific enthalpy, Btu/lb
Δh	specific enthalpy drop, Btu/lb
P	power
p	absolute vapor pressure, lb/sq in.
Q_s	heat supplied to cycle, Btu/hr
T	absolute temperature, $^{\circ}$R
w	weight-flow rate, lb/hr
δ	system-pressure loss, lb/sq in.
η	efficiency
ρ	density, lb/cu ft

Subscripts:

c	cycle
G	generator
id	ideal
P	recirculation pump
T	turbine
1	heater exit
2	separator inlet
3	separator exit, liquid
4	turbine inlet

5 turbine exit

6 condenser-radiator exit

7 condensate-pump exit

8 recirculation-pump inlet

9 recirculation-pump exit

METHOD OF ANALYSIS

As pointed out in the INTRODUCTION, this analysis was undertaken to determine the effect of heater-exit temperature, turbine-inlet temperature, turbine-exit to -inlet temperature ratio, fluid, and system-pressure loss on the recirculation-pump-power and radiator-area requirements for a modified Rankine power cycle system using flash vaporization.

A schematic diagram of a modified Rankine power cycle system employing flash vaporization is shown in figure 1(a), and the process is shown by a temperature-entropy diagram in figure 1(b). The process is described as follows. In the heater, the cycle working fluid is heated to T_1 (point 1 in fig. 1). The fluid expands across a restriction and some vaporization occurs. The two-phase mixture (point 2) enters a separator, where centrifugal force separates liquid from vapor. The saturated vapor (point 4) enters the turbine, while the liquid (point 3) is mixed with the discharge from the condensate pump. The vapor is expanded through the turbine and the expansion produces work to drive a generator. After leaving the turbine, the fluid (point 5) is condensed (point 6), its pressure is raised (point 7), and it is mixed with the liquid from the separator. The resulting liquid (point 8) is then raised to heater-inlet pressure (point 9) by the recirculation pump and returned to the heater.

Pump Power

The general expression for pump-input power, as applied to the recirculation pump, is

$$P_P = 5.423\times10^{-5} \frac{w_P(p_1 - p_2 + \delta)}{\eta_P \rho_8} \qquad \text{(kw)} \qquad (1)$$

The term $p_1 - p_2 + \delta$ denotes the total pressure drop in the flash loop and is equal to $p_9 - p_8$. The portion $p_1 - p_2$ represents the pressure drop due to the temperature change during the flash step, while δ, subsequently referred to as system-pressure loss, represents the frictional pressure drop plus the boiling-suppression head. For suppression of boiling in the heater, the heater-exit pressure must exceed the vapor pressure corresponding to the heater-exit temperature. This pressure difference is termed boiling-suppression head herein.

Since the quality of the fluid after flashing is low, T_8 does not differ much from T_2, and ρ_3 can be substituted for ρ_8 in equation (1). Expressing pump power as a fraction of generator output and introducing turbine weight flow into equation (1) yield

$$\frac{P_P}{P_G} = 5.423\times10^{-5} \frac{(p_1 - p_2 + \delta)}{\eta_P \rho_3} \frac{\dfrac{w_T}{P_G}}{\dfrac{w_T}{w_P}} \tag{2}$$

Turbine weight-flow rate per unit generator output is obtained as a function of fluid properties when turbine power is set equal to generator power:

$$w_T \, \Delta h_{id,T} \eta_T = 3415.2 \frac{P_G}{\eta_G} \qquad (\mathrm{Btu/hr})$$

Therefore,

$$\frac{w_T}{P_G} = \frac{3415.2}{\Delta h_{id,T} \eta_T \eta_G} \qquad (\mathrm{lb/(hr)(kw)}) \tag{3}$$

The ratio turbine- to pump-weight-flow rate (identical to the quality of fluid after flashing) is obtained from an enthalpy balance around the orifice and the separator. Heat losses are assumed negligible, so that $T_2 = T_3 = T_4$ and

$$w_P h_1 = w_T h_4 + (w_P - w_T) h_3 \qquad (\mathrm{Btu/hr})$$

consequently,

$$\frac{w_T}{w_P} = \frac{h_1 - h_3}{h_4 - h_3}$$

and with the enthalpy differences expressed in terms of fluid properties, the desired relation is obtained:

$$\frac{w_T}{w_P} = \frac{C_p(T_1 - T_2)}{\Delta H_{vap,2}} \tag{4}$$

Variation of liquid heat capacity with temperature is small enough that the mean heat capacity can be used. Substituting equations (3) and (4) into equation (2) yields

$$\frac{P_P}{P_G} = 0.1852 \frac{(p_1 - p_2 + \delta)\Delta H_{vap,2}}{\eta_P \rho_3 \ \Delta h_{id,T} \eta_T \eta_G C_p (T_1 - T_2)} \tag{5}$$

With a generator efficiency of 0.95, a turbine efficiency of 0.75, and a pump efficiency of 0.60 assumed, equation (5) becomes

$$\frac{P_P}{P_G} = 0.4332 \frac{(p_1 - p_2 + \delta)\Delta H_{vap,2}}{\rho_3 \ \Delta h_{id,T} C_p (T_1 - T_2)} \tag{6}$$

Equation (6) expresses the ratio of recirculation-pump power to generator power as a function of heater-exit temperature, turbine-inlet temperature, system-pressure loss, and the following temperature-dependent fluid properties: vapor pressure, heat of vaporization, liquid density, ideal specific turbine work, and liquid heat capacity. If these fluid properties are expressed as functions of temperature, equation (6) becomes a function of system temperatures alone for any given value of system-pressure loss.

Vapor pressure (ref. 5), heat of vaporization (ref. 6), liquid density (ref. 6), and liquid heat capacity (ref. 5) are plotted as a function of temperature for each fluid in figures 2(a) to (d). Ideal specific enthalpy drop in the turbine is a function of both turbine-inlet temperature and turbine-exit to -inlet temperature ratio for each fluid. Ideal specific enthalpy drop (from data presented in ref. 5) is presented in figure 2(e) as a function of turbine-inlet temperature for a turbine-exit to -inlet temperature ratio of 0.75, the value used for the bulk of the calculations.

Radiator Area

Specific radiator area is defined as the number of square feet of radiator surface required per kilowatt of net electrical power output.

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If a generator efficiency of 0.95 and a surface emissivity of 0.90 are assumed, the expression for specific radiator area (ref. 7) is

$$\frac{A_R}{P_G} = \frac{2.306}{(T_5/1000)^4}\left(\frac{1}{\eta_c} - 1\right) \qquad \text{(sq ft/kw)} \tag{7}$$

The assumptions used to derive equation (7) are:

(1) The waste heat is rejected from the working fluid at constant temperature T_5.

(2) The entire outer surface of the radiator is isothermal and the temperature is equal in magnitude to T_5.

(3) The effect of environmental sink temperature is negligible.

Cycle efficiency is defined as net power output divided by total heat supplied. Condensate-pump work is assumed to be negligible; therefore,

$$\eta_c = \frac{P_{T,net}}{Q_s} = \frac{P_T - P_P}{Q_s} \tag{8}$$

The terms appearing in cycle efficiency are evaluated as follows:

$$P_T = w_T(h_4 - h_5) = w_T\eta_T(h_4 - h_{5,id}) = w_T\eta_T\,\Delta h_{T,id} \qquad \text{(Btu/hr)}$$

$$P_P = w_P(h_9 - h_8) = \frac{144}{778}\,\frac{w_P}{\eta_P}\,\frac{(p_1 - p_2 + \delta)}{\rho_8} \qquad \text{(Btu/hr)}$$

$$Q_s = w_P(h_1 - h_9) \qquad \text{(Btu/hr)}$$

An enthalpy balance around the entire cycle yields

$$w_P(h_1 - h_9) + w_P(h_9 - h_8) = w_T(h_4 - h_5) + w_T(h_5 - h_6)$$

therefore,

$$Q_s = w_P(h_1 - h_9) = w_T(h_4 - h_6) - w_P(h_9 - h_8) \qquad \text{(Btu/hr)}$$

Since

$$(h_4 - h_6) = C_p(T_4 - T_6) + \Delta H_{vap,4}$$

and with the assumption that $T_6 = T_5$, the following relation is obtained:

$$Q_s = w_T\left[c_p(T_4 - T_5) + \Delta H_{vap,4}\right] - \frac{144}{778}\frac{w_P}{\eta_P}\frac{(p_1 - p_2 + \delta)}{\rho_8} \qquad \text{(Btu/hr)}$$

Substituting for P_T, P_P, and Q_s in equation (8) and letting $\eta_T = 0.7$, $\eta_P = 0.6$, and $\rho_8 = \rho_3$ give

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$$\eta_c = \frac{0.75\ \Delta h_{T,id} - 0.308\ \frac{w_P}{w_T}\frac{(p_1 - p_2 + \delta)}{\rho_3}}{c_p(T_4 - T_5) + \Delta H_{vap,4} - 0.308\ \frac{w_P}{w_T}\frac{(p_1 - p_2 + \delta)}{\rho_3}} \tag{9}$$

Specific radiator area for the Rankine cycle employing flash vaporization is therefore evaluated from equation (7), with η_c from equation (9), w_P/w_T from equation (4), and fluid properties from figure 2.

Recirculation-pump power as obtained from equation (6) and specific radiator area as obtained from equation (7) were investigated for the following conditions:

Fluid	sodium, potassium, or rubidium
Turbine-inlet temperature, T_4, $^{\circ}R$	1760 to 2960
Heater-exit temperature, T_1, $^{\circ}R$	T_4 to T_4 + 400
Turbine-exit to -inlet temperature ratio, T_5/T_4	0.6 to 0.9
System-pressure loss, δ, lb/sq in.	1 to 20

RESULTS OF ANALYSIS

The results of this analysis are presented in detail for an assumed system-pressure loss of 5 pounds per square inch; the effect of varying system-pressure loss from 1 to 20 pounds per square inch is also discussed.

Pump Power

Effect of heater-exit temperature. - The ratio of recirculation-pump power to generator power output is plotted against heater-exit temperature in figure 3 for each fluid and four turbine-inlet temperatures with a turbine-exit to -inlet temperature ratio of 0.75. Figure 3 is a working plot from which the effects of several variables can be determined. The discussion in this section is, however, restricted to the effect of heater-exit temperature. The effects of turbine-inlet temperature and fluid are discussed in subsequent sections.

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For each fluid and turbine-inlet temperature there is an optimum value of heater-exit temperature that yields a minimum recirculation-pump power. Recirculation-pump power, as seen from equation (1), is a function of the product of flash-loop flow rate and pressure drop. As heater-exit temperature decreases toward turbine-inlet temperature, the percentage of vaporization of the working fluid approaches zero, and the pump flow rate, and consequently pump power, must approach infinity in order to supply the required turbine flow. On the other hand, as heater-exit temperature increases, pressure drop will eventually increase faster than flow rate decreases and pump power will again increase. Consequently, there must be a value of heater-exit temperature for which recirculation-pump power is a minimum. The curves of figure 3 are fairly shallow near the minimums; it seems that a variation in heater-exit temperature of 50° R, or more in some cases, above or below the optimum does not significantly increase recirculation-pump power. The optimum flash-temperature drop (heater-exit temperature for minimum recirculation-pump power minus turbine-inlet temperature), presented in figure 4, decreases with an increase in turbine-inlet temperature. As mentioned previously, flash-temperature drop can be varied to some extent without significantly increasing recirculation-pump power; consequently, operation with a flash-temperature drop of 100° R yields minimum or very near minimum pump power for all temperature levels and fluids.

Effect of turbine-inlet temperature. - Minimum recirculation-pump power, as obtained from figure 3, for each fluid is plotted against turbine-inlet temperature in figure 5 for a turbine-exit to -inlet temperature ratio of 0.75. As seen from figure 5, minimum recirculation-pump power increases approximately tenfold as turbine-inlet temperature increases from 1760° to 2960° R. As turbine-inlet temperature increases, optimum flash-temperature drop (fig. 4) decreases, while the associated loop pressure drop increases, as can be determined from figure 2(a). It is primarily these two effects, as can be seen from equation (6), that cause the large increase in minimum recirculation-pump power.

Effect of fluid. - Figure 5 shows the effect of fluid on recirculation-pump power and also shows that the recirculation-pump power requirement increases as the fluid is changed from sodium to potassium to rubidium. The effect of fluid on pump power can be explained on the basis of equation (6). Pump power is directly proportional to loop pressure drop and heat of vaporization and inversely proportional to liquid density, ideal specific turbine work, liquid heat capacity, and flash-temperature drop. It is the interaction of these fluid effects as expressed by equation (6) that causes the fluid effect on pump power shown in figure 5.

For a sodium system with a 5-pound-per-square-inch system-pressure loss and a turbine-exit to -inlet temperature ratio of 0.75, minimum recirculation-pump power increases from 0.006 to 0.094 kilowatt per kilowatt of generator output as turbine-inlet temperature increases from 1760° to 2960° R. The corresponding pump powers for potassium and rubidium are 0.021 to 0.198 and 0.036 to 0.347, respectively.

With a design turbine-inlet temperature of 2560° R (a value believed both desirable and attainable in the near future), the total power output for a sodium system is 5 percent higher than that for a boiling-heat-exchanger cycle and for a rubidium system is 25 percent higher than that for a boiling-heat-exchanger cycle.

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Effect of turbine-exit to -inlet temperature ratio. - As seen from equation (6), the only term in the pump-power equation affected by turbine-exit to -inlet temperature ratio is the turbine-ideal-specific-enthalpy drop. Consequently, for any given fluid and heater-exit and turbine-inlet temperatures, a change in the turbine-exit to -inlet temperature ratio will merely displace the curves of figure 3 up or down and have no effect on the value of the optimum flash-temperature drop shown in figure 4. Recirculation-pump power for any given turbine-exit to -inlet temperature ratio divided by pump power at a turbine temperature ratio of 0.75 is shown in figure 6 as a function of turbine temperature ratio. Figure 6 was obtained from the data of reference 5 and is just the ratio of ideal-specific-enthalpy drops for a turbine-exit to -inlet temperature ratio of 0.75 to the given turbine temperature ratio. Within the accuracy of the thermodynamic data, this relation is independent of the fluids and turbine-inlet temperatures investigated. Figure 6 is merely a correction to be applied to the results in figures 3 and 5 to account for the effect of turbine-exit to -inlet temperature ratio for any given fluid and heater-exit and turbine-inlet temperatures. As shown in figure 6, recirculation-pump power increases from 0.6 to 2.5 times the power required at a temperature ratio of 0.75 as turbine temperature ratio increases from 0.6 to 0.9. Consequently, lowering the turbine temperature ratio can result in a significant reduction in required recirculation-pump power; however, this would be at the expense of increased radiator area, as is subsequently shown.

Effect of system-pressure loss. - The foregoing results were obtained with an assumed system-pressure loss of 5 pounds per square inch. Similar calculations were made for several other values of system-pressure loss in the range 1 to 20 pounds per square inch. For sodium with a turbine-inlet temperature of 2560° R, recirculation-pump power is plotted against heater-exit temperature in figure 7 with system-pressure loss as a parameter. For the other fluids and turbine-inlet temperatures, the corresponding sets of curves are quite similar to those shown in figure 7. Since system-pressure loss depends on system design rather than on basic cycle considerations and the general effect of this loss can be determined from the one case shown, it is unnecessary to present

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the curves for the other fluids and turbine-inlet temperatures. As shown in figure 7, the effects of increasing system-pressure loss are a significant increase in optimum flash-temperature drop and a small absolute increase in minimum recirculation-pump power. For the case shown, optimum flash-temperature drop increases by 150° R and minimum recirculation-pump power increases by 0.016 kilowatt per kilowatt of generator output as system-pressure loss increases from 1 to 20 pounds per square inch. The increases for the other fluids and turbine-inlet temperatures are of a comparable magnitude.

Radiator Area

Effect of turbine-exit to -inlet temperature ratio. - Specific radiator area is plotted against turbine-exit to -inlet temperature ratio in figure 8 for each fluid with turbine-inlet temperatures of 2160° and 2560° R. Flash-temperature drops of 50°, 100°, and 150° R were assumed for each fluid and turbine-inlet temperature; however, since the resulting curves coincided within 1 percent, single curves are shown to represent the results obtained with flash-temperature drops in the 50° to 150° R range. As shown in figure 8, minimum specific radiator area occurs at turbine-exit to -inlet temperature ratios of 0.74 to 0.77, the exact value depending on the fluid and turbine-inlet temperature. Minimum specific radiator area for a modified Rankine cycle system employing a boiling heat exchanger and having a turbine efficiency of 0.75 occurs at a turbine temperature ratio of about 0.77 and is essentially independent of fluid (ref. 7); thus, the variation in turbine temperature ratio for minimum area and the fluid effect shown in figure 8 are due to recirculation-pump-power requirements. As recirculation-pump power increases, specific radiator area increases and optimum area occurs at lower values of turbine temperature ratio. However, even with a recirculation-pump power as high as 0.21 kilowatt per kilowatt of generator power output (rubidium at $T_4 = 2560^\circ$ R) the shift in optimum turbine temperature ratio is quite small (to 0.74). Consequently, subsequent calculations are made with a turbine-exit to -inlet temperature ratio of 0.75.

Effect of heater-exit and turbine-inlet temperatures. - In studying specific-radiator-area requirements as a function of heater-exit and turbine-inlet temperatures, there are two cases to be considered. In one case the turbine is assumed to be the temperature-limited component in the flash loop, and the heater-exit temperature can take on any reasonable value above the limiting turbine-inlet temperature. In the other case a limiting temperature is assumed for the entire flash loop; consequently, the heater-exit temperature must be set equal to this limiting temperature. The two cases yield results of a little different nature.

The case of limiting turbine temperature is considered first. For any given fluid and turbine-inlet temperature, a plot of specific radiator area against heater-exit temperature would show a minimum radiator area at the same flash-temperature drop (fig. 4) required for minimum recirculation-pump power. This can be seen by examination of the equations for specific radiator area and cycle efficiency. For a given turbine-exit temperature, equation (7) shows that radiator area is minimized where cycle efficiency is maximized. Examination of equation (9) shows that cycle efficiency is maximized when recirculation-pump power, the last term in both numerator and denominator, is minimized; therefore, specific radiator area is also minimized at the flash-temperature drop required for minimum recirculation-pump power.

Specific radiator area, minimized with respect to flash-temperature drop, is plotted against turbine-inlet temperature in figure 9(a) for a turbine-exit to -inlet temperature ratio of 0.75. For comparison, specific radiator area for a modified Rankine cycle employing a boiling heat exchanger is shown. As mentioned previously, minimum specific radiator area for this type of cycle is essentially independent of fluid. Minimum specific radiator area decreases from a range of 3.3 to 3.6 to a range of 0.41 to 0.63 square foot per kilowatt of generator output as turbine-inlet temperature increases from 1760° to 2960° R. This decrease in radiator area is due to the increasing radiator temperature that results from the increasing system temperature level. The increase in specific radiator area for the flash-boiling cycles over that required for the boiling-heat-exchanger cycle is due to the required recirculation-pump power. Increased recirculation-pump power results in lower cycle efficiency which, in turn, yields a higher specific radiator area.

For the case of limiting system temperature, a heater-outlet temperature of 2600° R is chosen for examination. Specific radiator area is plotted against turbine-inlet temperature in figure 9(b) for a turbine-exit to -inlet temperature ratio of 0.75. For comparison, the point representing a modified Rankine cycle with a boiling heat exchanger for a maximum system temperature (turbine inlet) of 2600° R is shown. The optimum flash-temperature drop for minimum specific radiator area (15° to 30° R) is smaller than that for minimum recirculation-pump power (~100° R). As flash-temperature drop decreases below the value for minimum recirculation-pump power, cycle efficiency decreases because of increasing pump power, while radiator temperature increases. As long as the effect of increasing radiator temperature more than offsets the effect of increasing pump power, radiator area will decrease; however, at small flash-temperature drops recirculation-pump power increases very rapidly (fig. 3), and radiator area will start to increase as turbine-inlet temperature approaches the fixed maximum heater-exit temperature. The difference in required radiator area between operation at the flash-temperature drop for minimum recirculation-pump power and that for minimum specific radiator area is less than 10 percent.

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Effect of system-pressure loss. - As with the results for recirculation-pump power, the foregoing results for specific radiator area were obtained with an assumed system-pressure loss of 5 pounds per square inch. Similar calculations can be made for other values of system-pressure loss; however, this was considered unnecessary for the following reasons. System-pressure loss affects specific radiator area only because it affects cycle efficiency through a change in recirculation-pump power. As pointed out previously, increasing system-pressure loss from 1 to 20 pounds per square inch does not significantly affect the absolute value of minimum recirculation-pump power. Furthermore, as can be seen from equation (9), recirculation-pump power is a term that is subtracted from both the numerator and the denominator of the expression for cycle efficiency; consequently, a small change in recirculation-pump power results in an even smaller change in cycle efficiency. As a result, an increase in system-pressure loss from 1 to 20 pounds per square inch will increase specific radiator area by only 5 percent for the worst case investigated (rubidium at $T_4 = 2960^\circ$ R).

SUMMARY OF RESULTS

This investigation was undertaken to determine the effect of heater-exit temperature, turbine-inlet temperature, turbine-exit to -inlet temperature ratio, fluid, and system-pressure loss (frictional loss plus boiling-suppression head) on the recirculation-pump-power and radiator-area requirements for a modified Rankine power cycle employing flash vaporization. Sodium, potassium, and rubidium were investigated as working fluids, with turbine-inlet temperature ranging from 1760° to 2960° R and heater-exit temperature ranging from 0° to 400° R above turbine-inlet temperature. Turbine-exit to -inlet temperature ratio was varied from 0.6 to 0.9 and system-pressure loss was varied from 1 to 20 pounds per square inch. The pertinent results are summarized as follows:

1. For each fluid, turbine-inlet temperature, and system-pressure loss, there is an optimum value of heater-exit temperature that yields a minimum recirculation-pump power. Optimum flash-temperature drop decreases as turbine-inlet temperature increases and system-pressure loss decreases and is independent of turbine-exit to -inlet temperature ratio. For a system-pressure loss of 5 pounds per square inch, operation with a flash-temperature drop of 100° R yields minimum or very near minimum recirculation-pump power for all turbine-inlet temperatures and fluids studied.

2. Minimum recirculation-pump power increases with an increase in turbine-inlet temperature, turbine-exit to -inlet temperature ratio, and system-pressure loss. A sodium system requires the lowest recirculation-pump power, potassium higher power, and rubidium the highest. For a sodium system with a 5-pound-per-square-inch system-pressure loss and a

turbine-exit to -inlet temperature ratio of 0.75, minimum recirculation-pump power increases from 0.006 to 0.094 kilowatt per kilowatt of generator output as turbine-inlet temperature increases from 1760° to 2960° R. The corresponding pumping powers are 0.021 to 0.198 for potassium and 0.036 to 0.347 for rubidium. As turbine-exit to -inlet temperature ratio increases from 0.6 to 0.9, recirculation-pump power increases from 0.6 to 2.5 times the power required at a turbine temperature ratio of 0.75. The increase in minimum recirculation-pump power with increasing system-pressure loss is quite small in the range studied.

3. For each fluid, turbine-inlet temperature, and heater-exit temperature, there is an optimum value of turbine-exit to -inlet temperature ratio that yields a minimum specific radiator area. Optimum turbine temperature ratio decreases from 0.77 for zero recirculation-pump power (boiling-heat-exchanger cycle) to 0.74 for a recirculation-pump power of 0.21 kilowatt per kilowatt of generator output. The curves are fairly shallow near the minimums, and a turbine-exit to -inlet temperature ratio of 0.75 can be assumed in order to find minimum radiator area for all cases.

4. If the turbine is considered to be the temperature-limited component in the system, the optimum flash-temperature drop for minimum specific radiator area is the same as that required for minimum recirculation-pump power. If the heater is considered to be the temperature-limited component in the system, the optimum flash-temperature drop for minimum radiator area is quite a bit less than that required for minimum recirculation-pump power. The difference in required radiator area between operation at the flash-temperature drop for minimum recirculation-pump power and that for minimum specific radiator area is, however, less than 10 percent.

5. Turbine-inlet temperature is the prime factor in determining minimum specific radiator-area requirements. Minimum specific radiator area decreases from 3.3 to 3.6 square feet per kilowatt to 0.41 to 0.63 square foot per kilowatt as turbine-inlet temperature increases from 1760° to 2960° R. The increase in radiator area over that required for a boiling-heat-exchanger cycle is proportional to the increase in total power output that is required for operation of the recirculation pump.

6. This study shows that, with a design turbine-inlet temperature of 2560° R (a value believed both desirable and attainable in the near future), the total power output and radiator-area requirements for a sodium system are 5 percent higher than those for a boiling-heat-exchanger cycle and that the requirements for a rubidium system are 25 percent higher than those for a boiling-heat-exchanger cycle. The choice

of working fluid is, therefore, an important factor in the consideration of a modified Rankine cycle employing flash vaporization.

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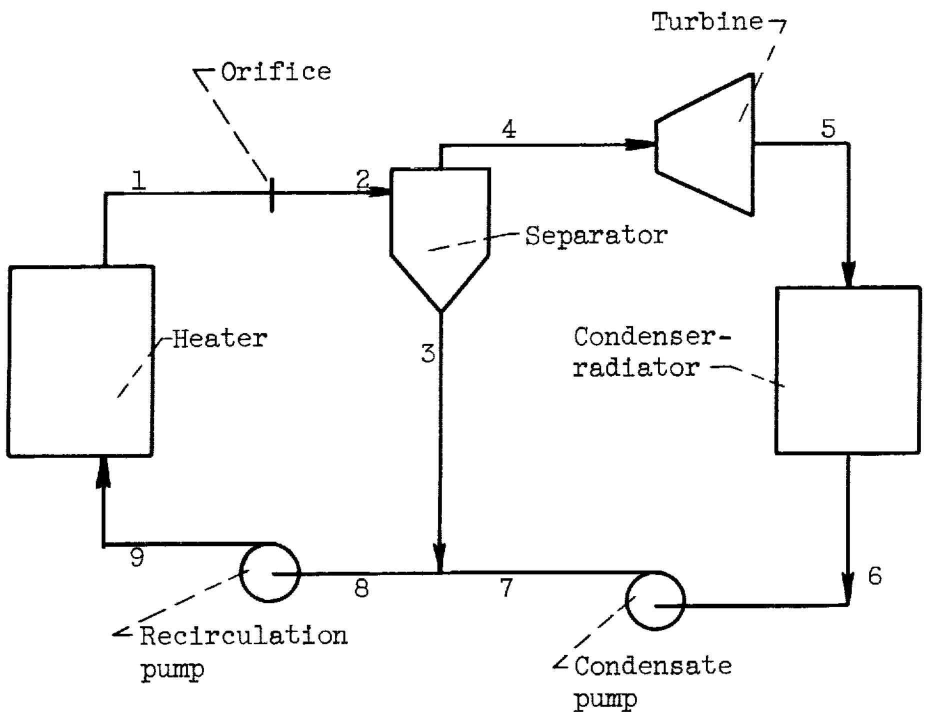
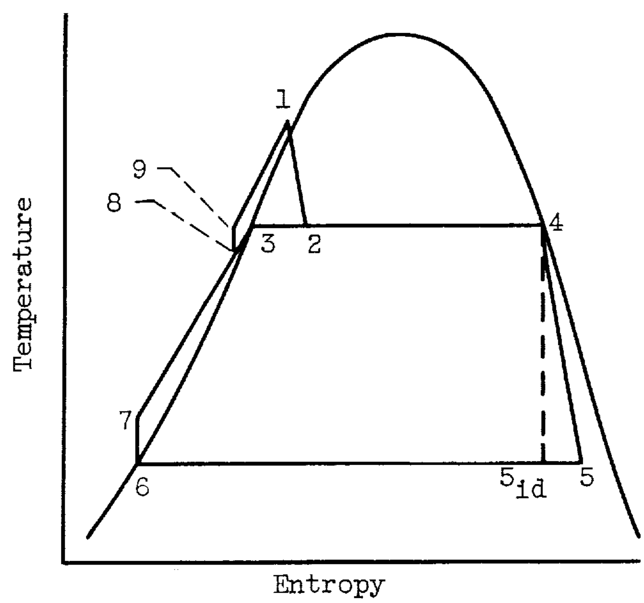


(a) Schematic diagram of system.



(b) Temperature-entropy diagram.

Figure 1. - Rankine power cycle employing flash vaporization.

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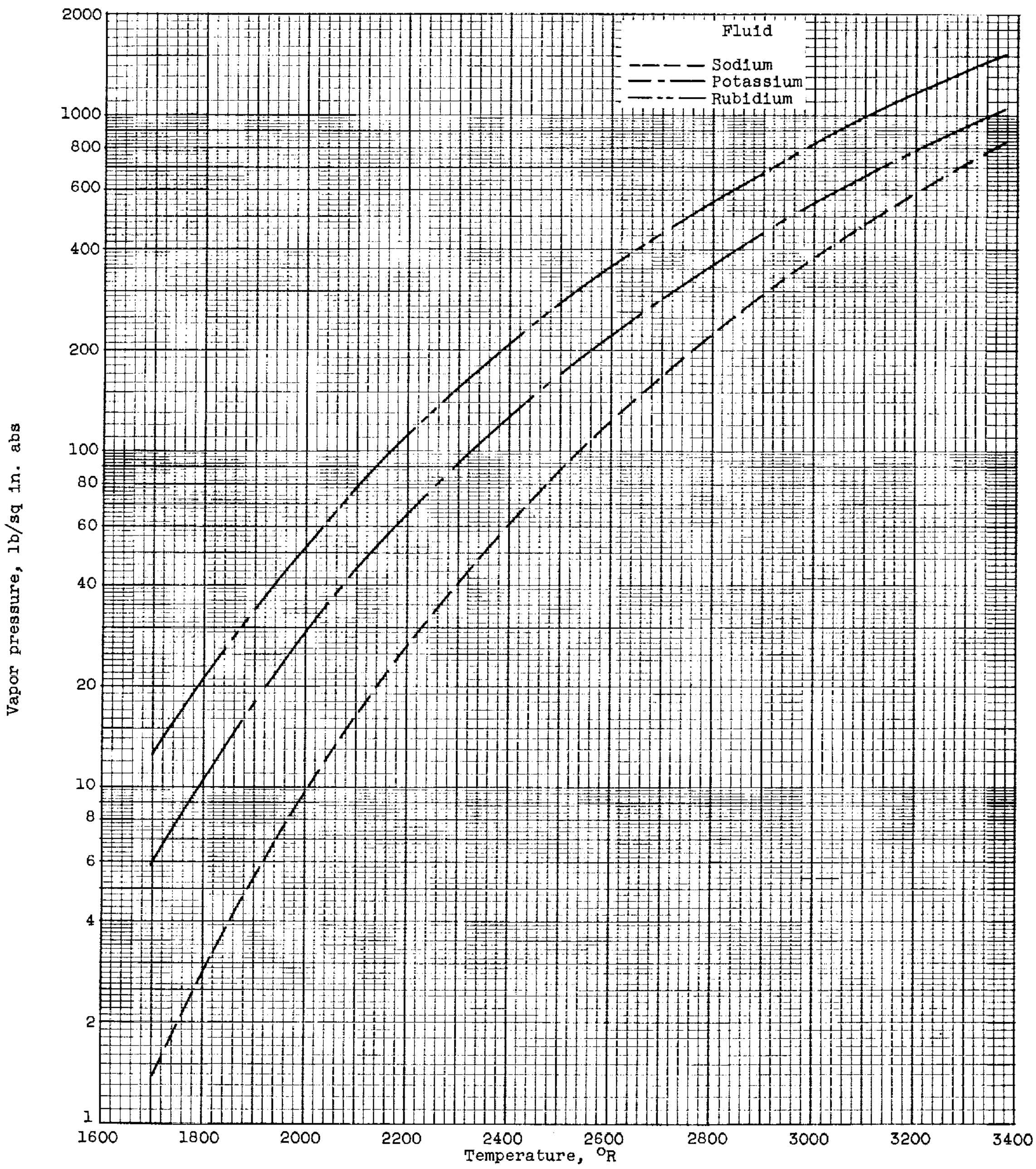
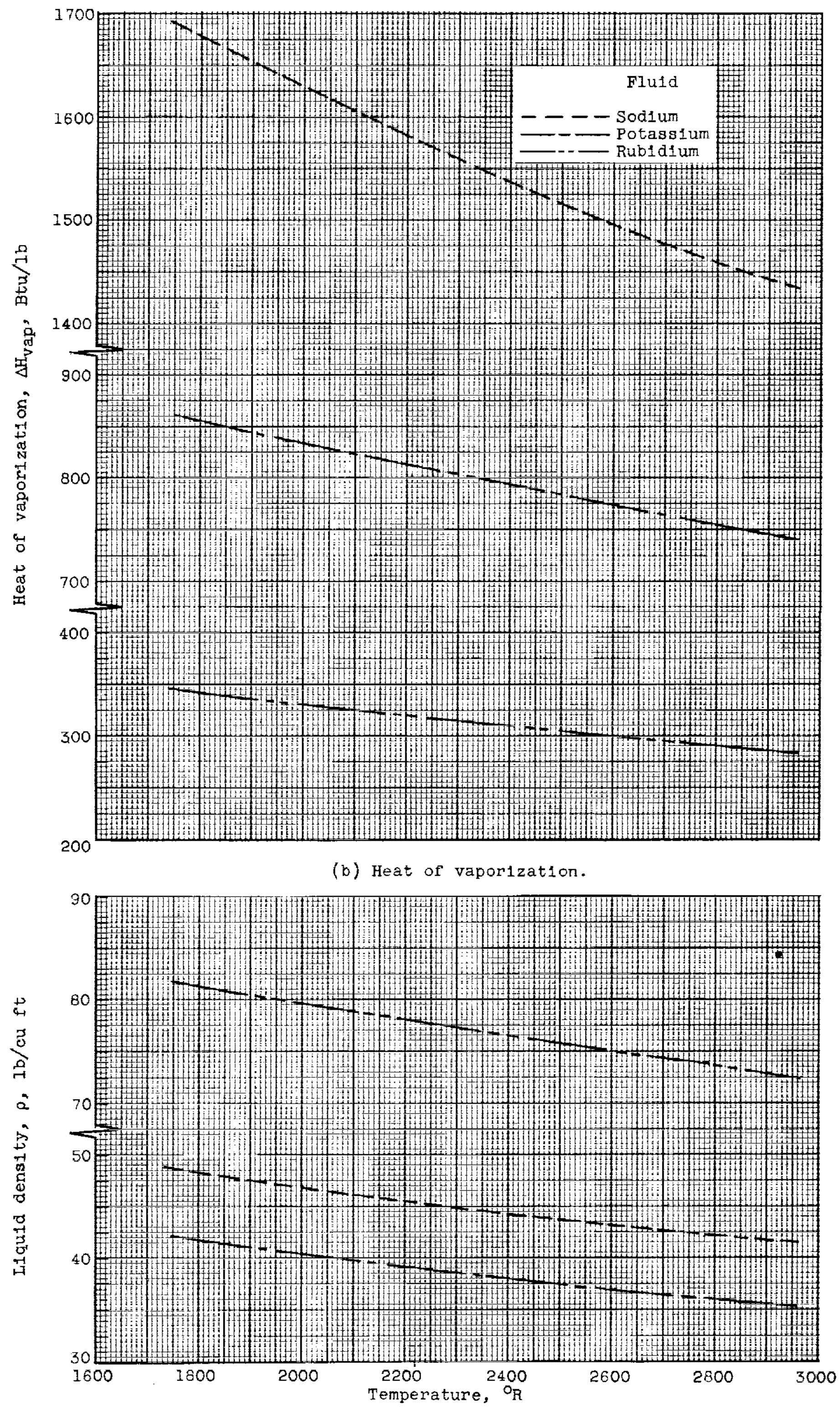


(a) Vapor pressure.

Figure 2. - Effect of temperature on fluid properties.



(c) Liquid density.

Figure 2. - Continued. Effect of temperature on fluid properties.

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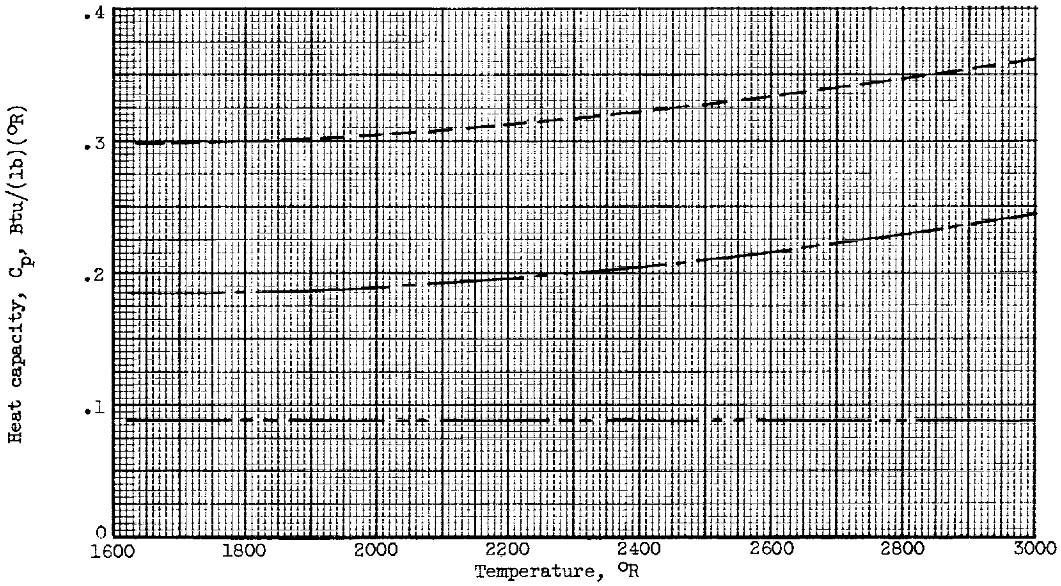
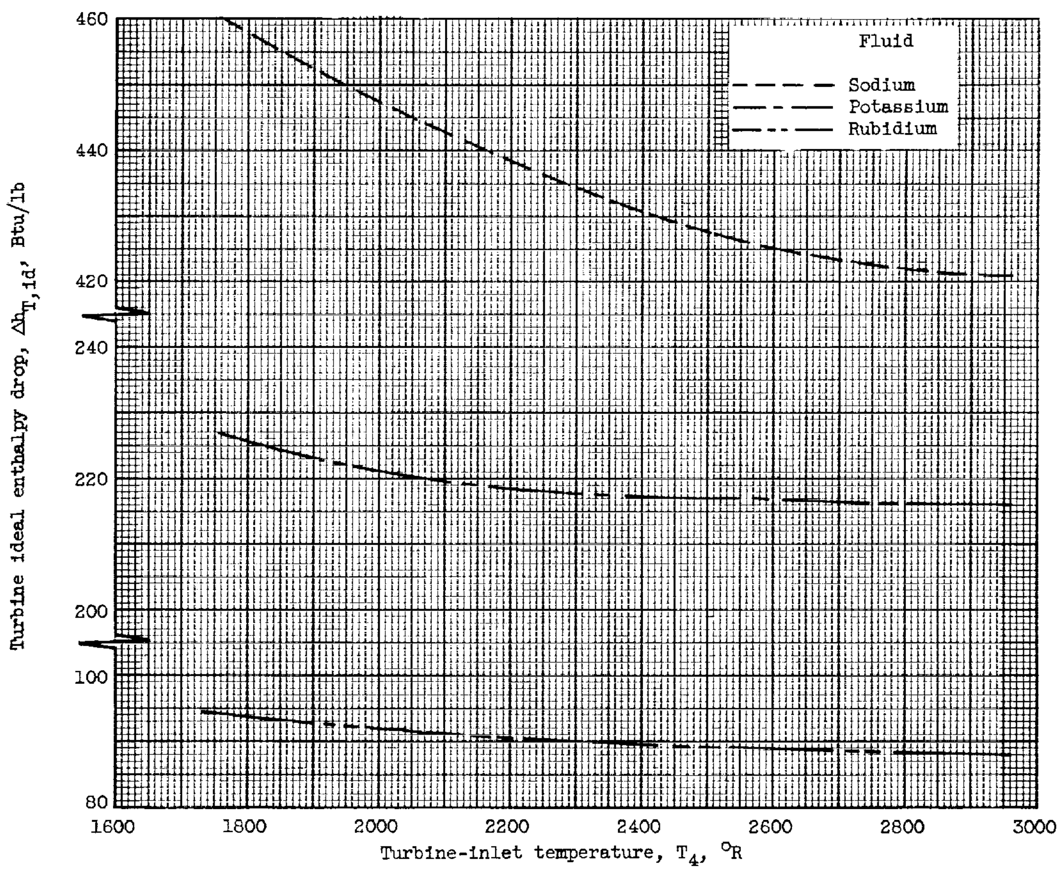


(d) Heat capacity.



(e) Turbine ideal enthalpy drop. Turbine-exit to -inlet temperature ratio, 0.75.

Figure 2. - Concluded. Effect of temperature on fluid properties.

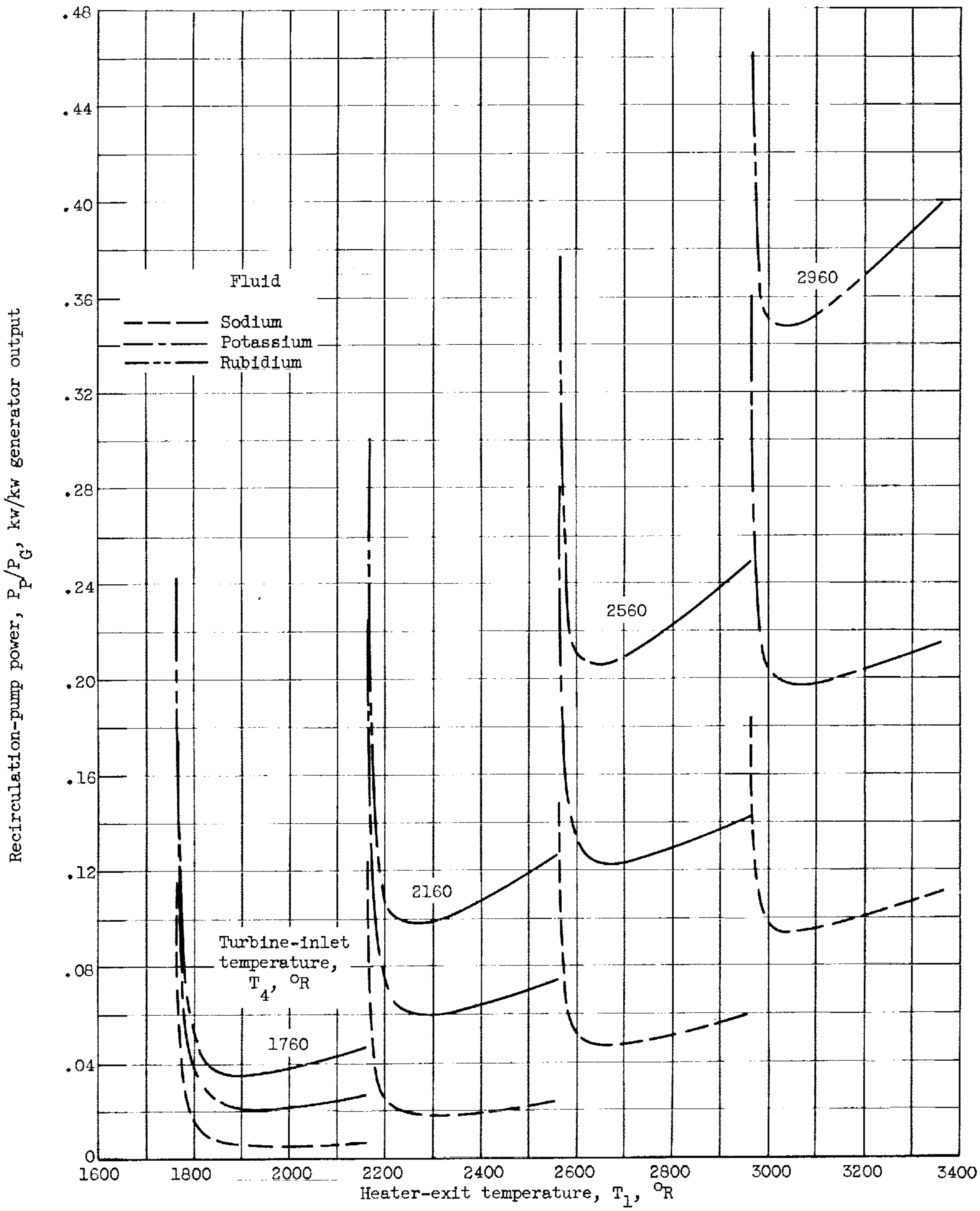


Figure 3. - Effect of heater-exit and turbine-inlet temperatures on recirculation-pump power. System-pressure loss, 5 pounds per square inch; turbine-exit to -inlet temperature ratio, 0.75.

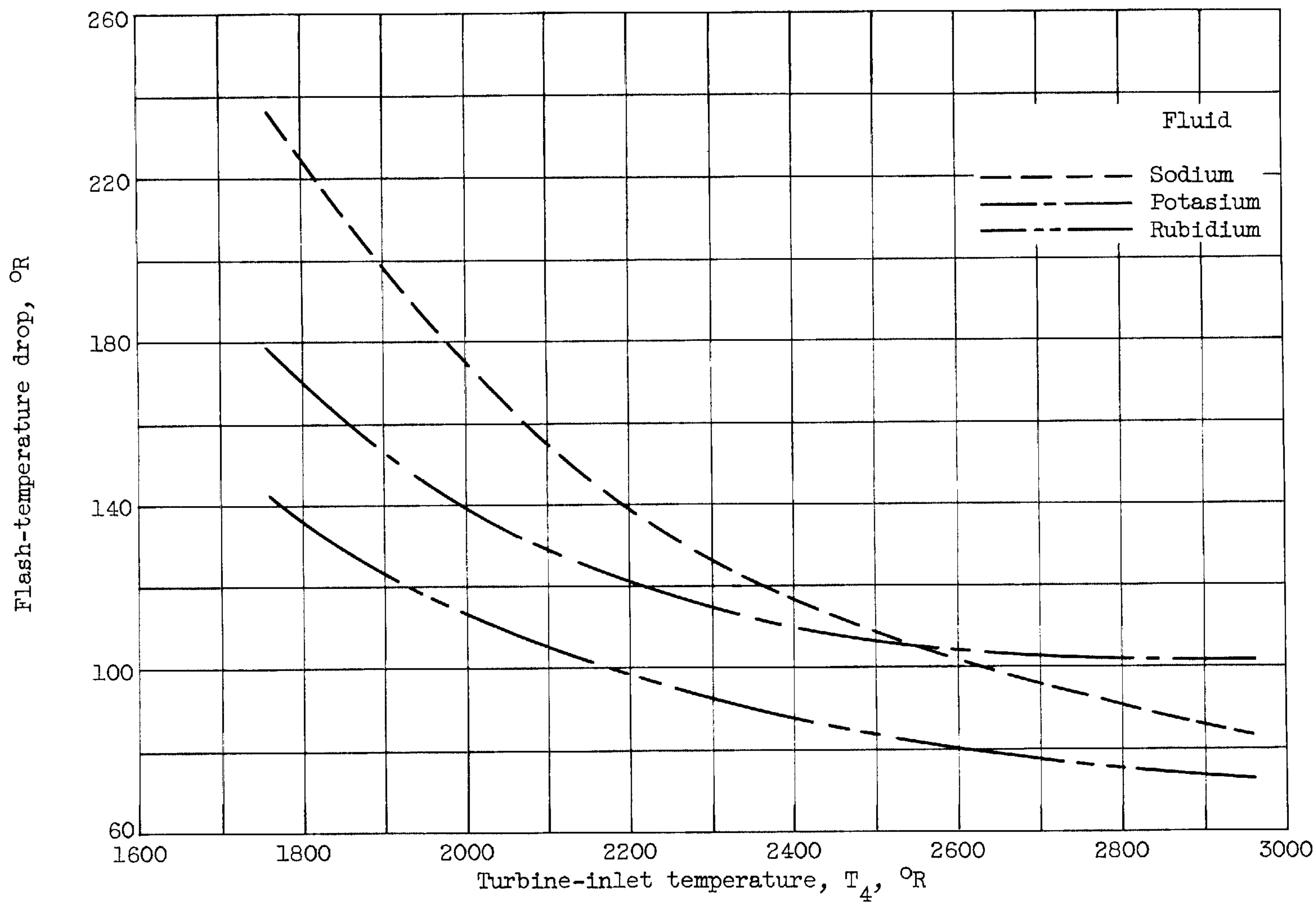


Figure 4. - Effect of turbine-inlet temperature on flash-temperature drop for minimum recirculation-pump power. System-pressure loss, 5 pounds per square inch.

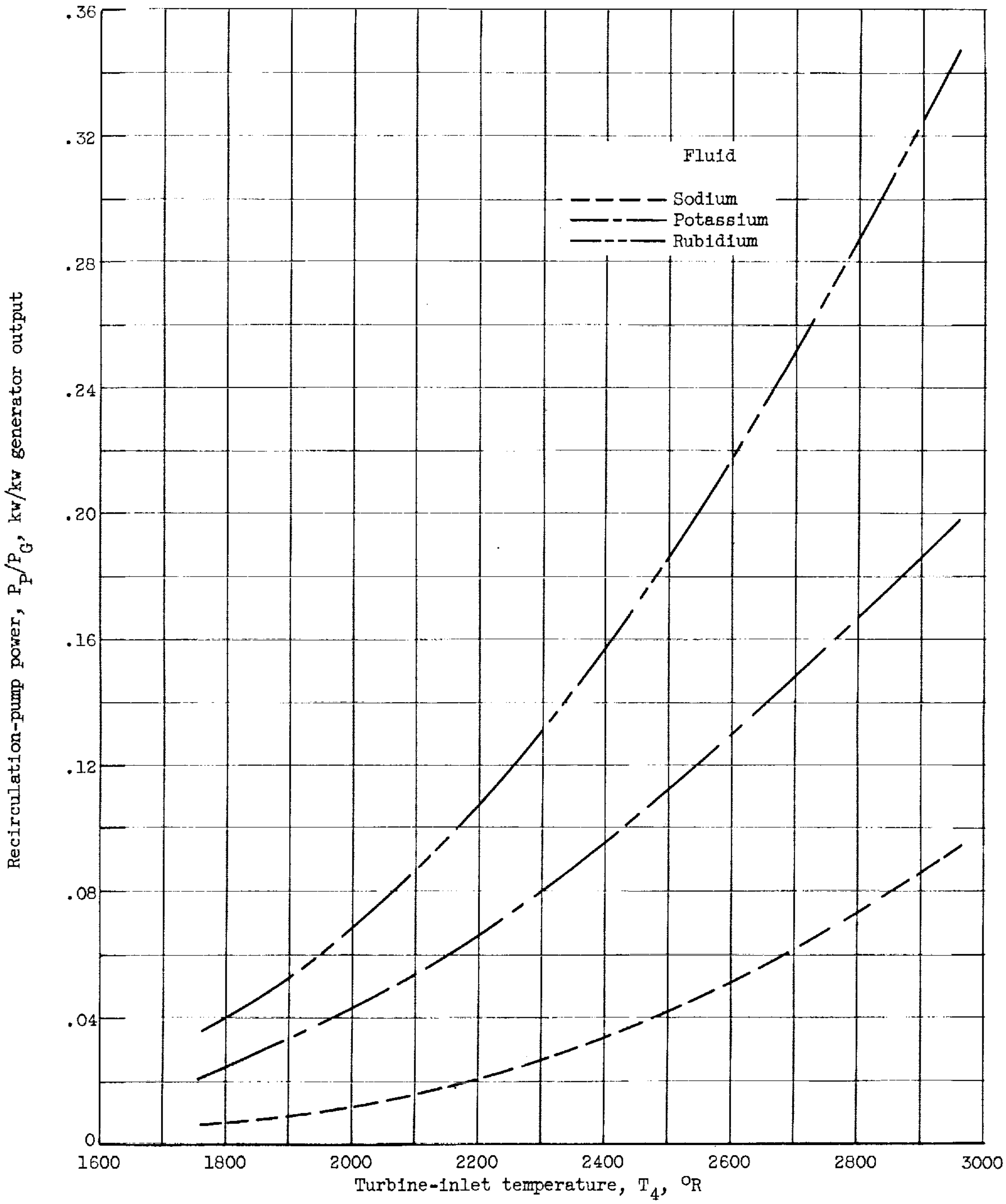


Figure 5. - Effect of turbine-inlet temperature on minimum recirculation-pump power. Turbine-exit to -inlet temperature ratio, 0.75; system-pressure loss, 5 pounds per square inch.

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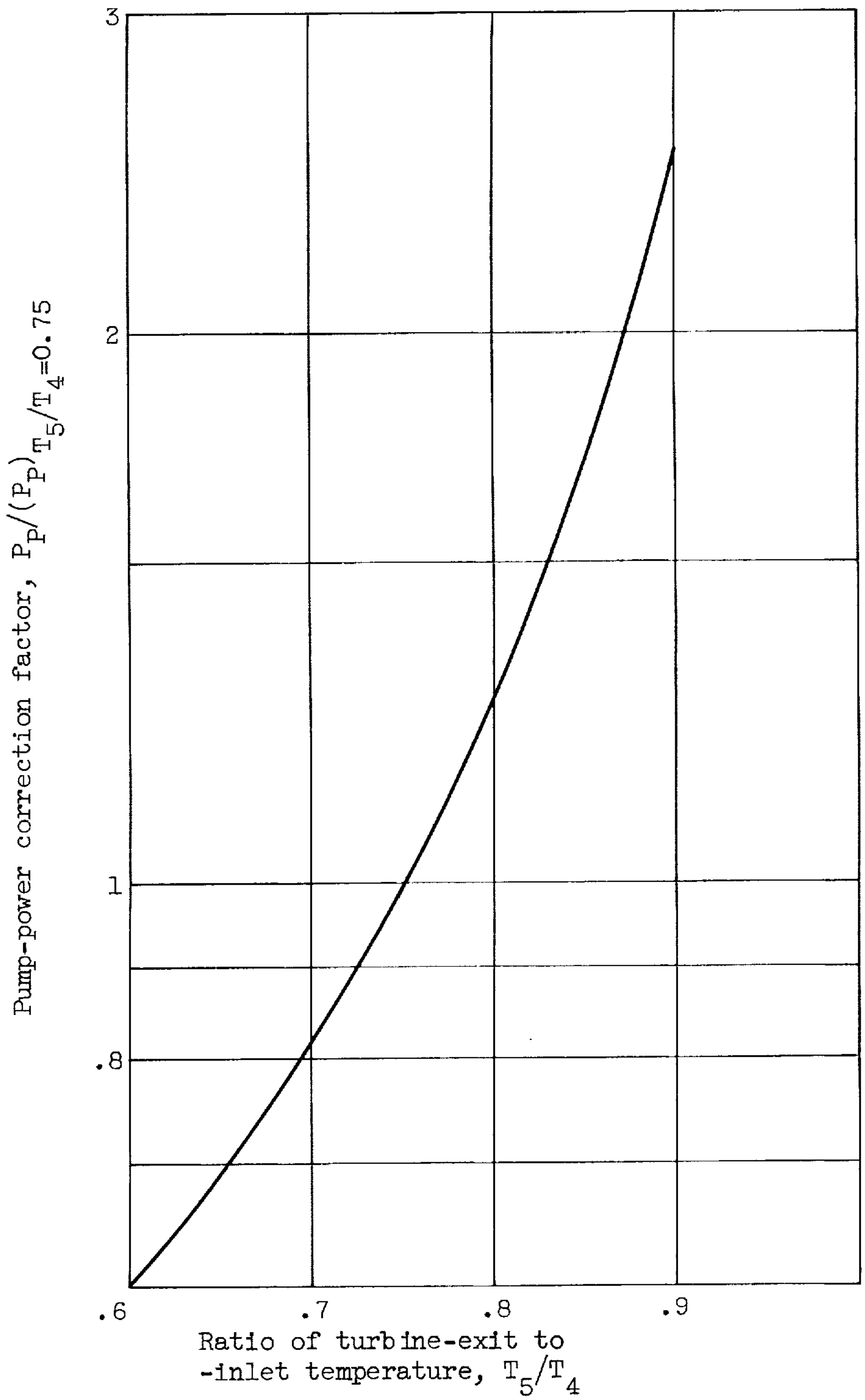


Figure 6. - Effect of turbine temperature ratio on recirculation-pump power.

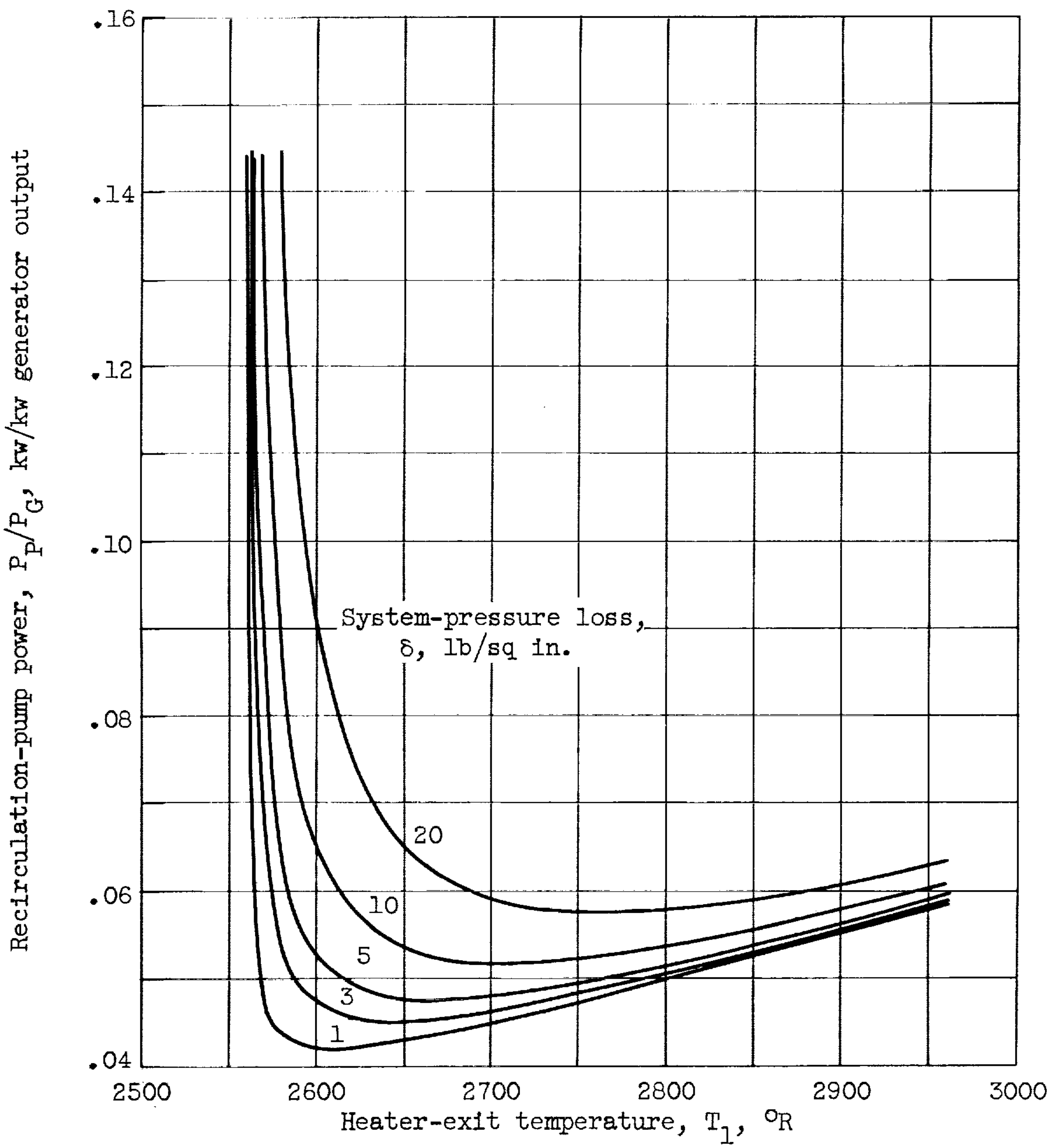


Figure 7. - Effect of system-pressure loss on recirculation-pump power. Fluid, sodium; turbine-inlet temperature, 2560° R; turbine-exit to -inlet temperature ratio, 0.75.

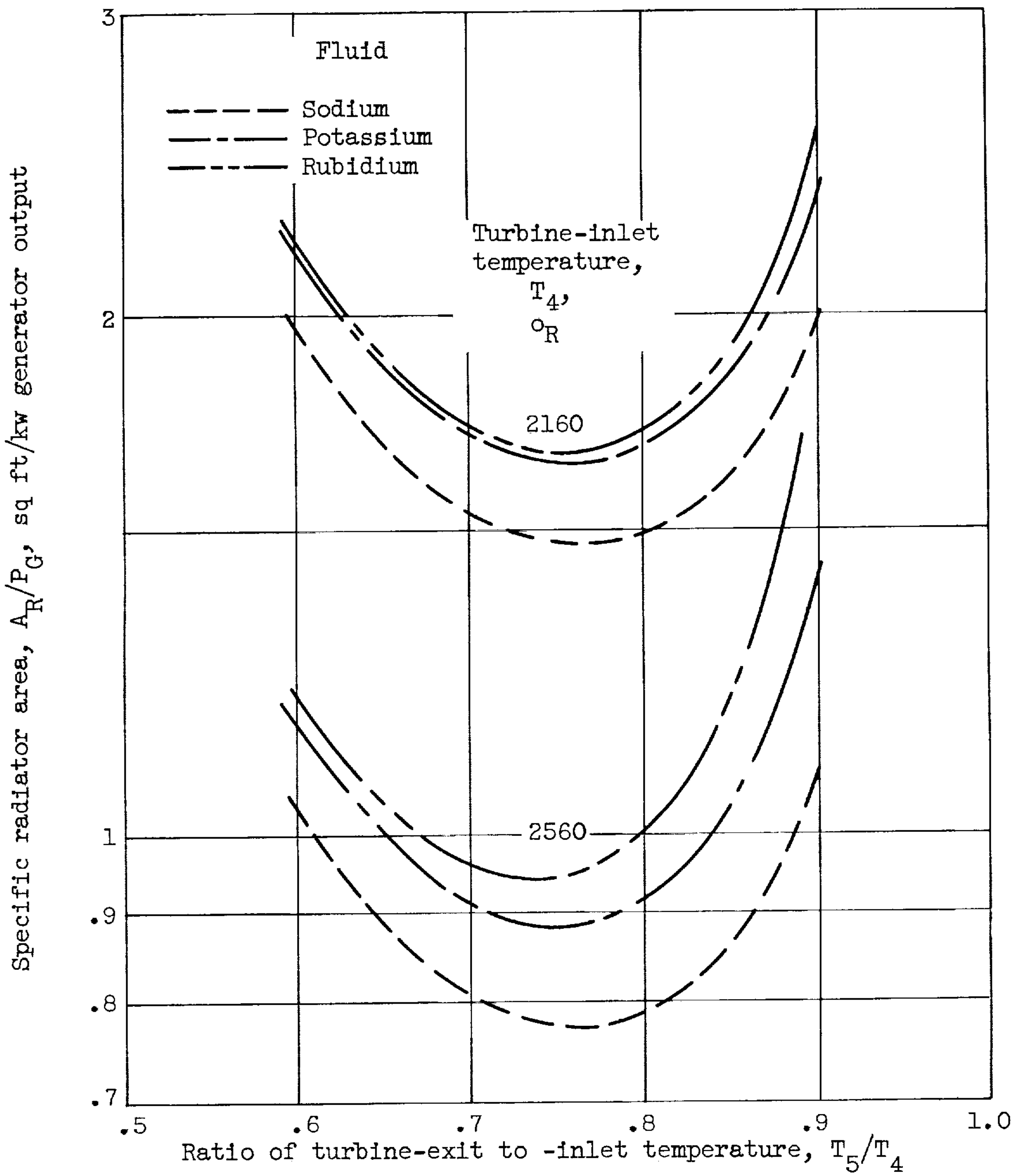
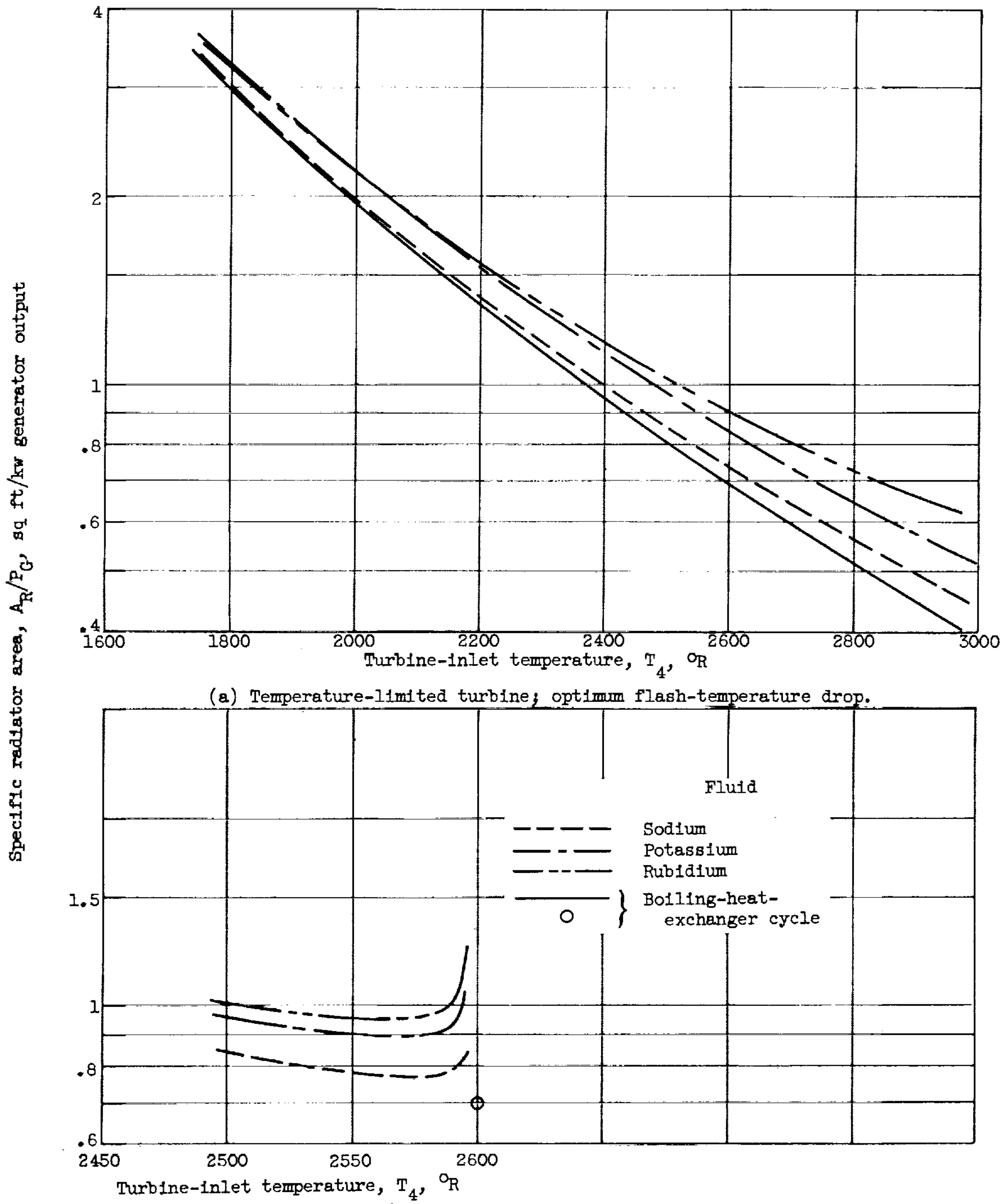


Figure 8. - Effect of turbine temperature ratio on specific radiator area. System-pressure loss, 5 pounds per square inch.



(a) Temperature-limited turbine; optimum flash-temperature drop.

(b) Temperature-limited heater; heater-exit temperature, 2600° R.

Figure 9. - Effect of turbine-inlet temperature on specific radiator area. Turbine-exit to -inlet temperature ratio, 0.75; system-pressure loss, 5 pounds per square inch.

NASA TN D-1283
National Aeronautics and Space Administration.
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(NASA TECHNICAL NOTE D-1283)

An investigation was made to determine the effect of heater-exit temperature, turbine-inlet temperature, turbine-exit to -inlet temperature ratio, fluid, and system-pressure loss on the recirculation-pump-power and radiator-area requirements for a modified Rankine power cycle employing flash vaporization. Sodium, potassium, and rubidium were investigated as working fluids, with turbine-inlet temperature ranging from 1760° to 2960° R and heater-exit temperature ranging from 0° to 400° R above turbine-inlet temperature. Turbine-exit to -inlet temperature ratio was varied from 0.6 to 0.9, and system-pressure loss was varied from 1 to 20 lb/sq in.

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